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By

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A STUDY OF HEAT TRANSFER IN A CIRCULAR CHANNEL

Kh. A. Barlybayev and S. V. Bukhman

Flow of a heat-transfer agent in circular channels formed by two concentric cylinders is widely used in modern heat-transfer apparatus. As a rule, flow of a fluid in such channels is turbulent. The analytical solution of the problem of heat transfer in a laminar regime, however, is of considerable interest. As in other cases, such a solution, which rests on physically strict equations, allows the mechanism of the phenomenon to be revealed and a number of qualitative laws which are preserved in turbulent flow to be established. Within the framework of the problem in question, let us first of all consider the effect of the geometry of the channel (the ratio of the diameters of the outer and inner pipes forming the circular channel), which does not reduce, naturally, merely to the introduction of an equivalent diameter into the formula for calculation of heat transfer.

Statement of the Problem

Let us examine the steady-state, hydrodynamically and thermally stabilized flow of an incompressible fluid with constant physical

parameters along a circular channel formed by two coaxial cylindrical pipes (the diagram and position of the X and r axes are shown in Fig. 1). The radii of the inner and outer pipes will be denoted by R_1 and R_2, respectively.

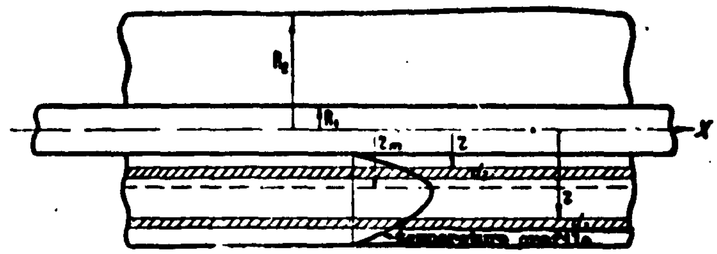

Fig. 1. Diagram of circular channel.

Let the circular space between the inner pipe and the cylindrical surface of radius r_m (Fig. 1) on which the maximum temperatures lie be section 1, and let the space between the outer pipe and this same cylindrical surface be section 2. Let us isolate in these sections circular layers with radius r and thickness dr. The difference between the amounts of heat introduced and removed by a fluid in a section of length dx in the region from r_m to r let us call Q. Since on the cylindrical surface of radius r_m the derivative of the temperature is equal to zero $\left(\frac{\partial t}{\partial r}\right)_m = 0$, then a quantity of heat Q will be transmitted through the layer of thickness dr. In this case the following equation holds:

$$\int_{r}^{r_m} 2\pi\rho C_p \frac{\partial t}{\partial x} U(r)\, r\, dr = 2\pi\lambda r \frac{\partial t}{\partial r}, \qquad (1)$$

where ρ, C_p and λ are the density, specific heat and thermal conductivity of the fluid, and U(r) is the flow rate of the fluid.

With steady-state motion, from the condition of constancy of heat flows through the walls of a channel it follows that the tem-

perature of the walls of the channel and the temperature of the fluid vary along the channels according to a linear law. Therefore, the change in temperature of the fluid in a channel of length dx can be determined from the heat balance for the entire stream

$$\pi (R^2_2 - R^2_1)\, \overline{U} \gamma C_p \frac{\partial t}{\partial x} = 2\pi (q_1 R_1 + q_2 R_2).$$

Hence

$$\frac{\partial t}{\partial x} = \frac{2(q_1 R_1 + q_2 R_2)}{(R^2_2 - R^2_1)\, \overline{U} \gamma C_p}. \qquad (2)$$

where $\overline{U}$ is the average velocity of the fluid in the channel, q_1 and q_2 the heat flows through the inner and outer walls of the channel.

Substituting the value of $\frac{\partial t}{\partial x}$ into (1), we obtain an equation which allows us to find the temperature field

$$\int_r^{r_m} \frac{2(q_1 R_1 + q_2 R_2)}{(R^2_2 - R^2_1)\, \overline{U}}\, U(r) \cdot r dr = r\lambda \frac{\partial t}{\partial r}. \qquad (3)$$

The radius of the maximum of the temperature profile r_m can be found from the condition that the heat introduced by the fluid into section 1 or 2 will be transmitted to the inner and outer pipes forming the channel, respectively. This condition for section 1 has the form:

$$\int_{R_1}^{r_m} 2\pi\gamma C_p\, U(r) \frac{\partial t}{\partial x}\, r dr = 2\pi R_1 q_1. \qquad (4)$$

Substituting the value of $\frac{\partial t}{\partial x}$ from (2) into this expression, we obtain

$$2\int_{R_1}^{r_m} \frac{q_1 R_1 + q_2 R_2}{(R^2_2 - R^2_1)\, \overline{U}}\, U(r)\, r dr = R_1 q_1. \qquad (5)$$

Expression (5) is the equation necessary for determining r_m (in the limit of the integral). In order to solve it, it is necessary to know

the dependence of velocity upon the radius.

Fluid Flow with a Constant Velocity

Let us consider the solution of the problem for the case of a uniform velocity distribution along the cross section of the channel ($U(r) = \bar{U} = \text{const}$). Under this assumption, the solutions of Eq. (4) separately for sections 1 and 2 have the form:

$$\vartheta_1 \equiv t - t_{1w} = \frac{q_1 R_1 + q_2 R_2}{\lambda (R_2^2 - R_1^2)} \left(r_m^2 \ln \frac{r}{R_1} - \frac{r^2}{2} + \frac{R_1^2}{2} \right). \tag{6}$$

$$\vartheta_2 \equiv t - t_{2w} = \frac{q_1 R_1 + q_2 R_2}{\lambda (R_2^2 - R_1^2)} \left(r_m^2 \ln \frac{r}{R_2} - \frac{r^2}{2} + \frac{R_2^2}{2} \right). \tag{7}$$

where t_{1w} and t_{2w} are the temperatures of the inner and outer pipes, respectively. Here we use the boundary conditions $t = t_{1w}$ at $r = R_1$ and $t = t_{2w}$ at $r = R_2$.

The temperatures of the walls of the inner and outer pipes are interlinked by the relationship

$$t_{1w} = 1_{2w} + \frac{q_1 R_1 + q_2 R_2}{\lambda (R_2^2 - R_1^2)} \left(r_m^2 \ln \frac{R_1}{R_2} + \frac{R_2^2}{2} - \frac{R_1^2}{2} \right). \tag{8}$$

If in Eq. (6) the temperature of the inner pipe (t_{1w}) is expressed in terms of the temperature of the outer pipe (t_{2w}), according to Eq. (8), then for the temperature at any point in the channel we have the equation

$$\frac{t - t_{2w}}{t_m - t_{2w}} = \frac{2 r_m^2 \ln \frac{r}{R_2} - r^2 + R_2^2}{2 r_m^2 \ln \frac{r_m}{R_2} - r_m^2 + R_2^2}, \tag{9}$$

where t_m is the maximum temperature, corresponding to the radius r_m.

The value of r_m entering into this equation is found from (5)

$$\frac{r_m}{R_1} = \sqrt{\frac{1 + q_{21} R_{12}}{1 + q_{12} R_{21}}}.^{*} \tag{10}$$

Under the condition of constancy of heat flows through the inner and outer walls of the channel, the value of r_m is defined as the geometric mean of the radii of the tubes forming the channel, and has the form:

$$r_m = \sqrt{R_1 R_2}.$$

Figure 2 shows the dimensionless-temperature profiles constructed according to Eq. (9) for a channel with $R_{21} = 2$ at various ratios of heat flows through the inner and outer pipes, and also temperature profiles for various circular channels ($R_{21} = 2, 10, 100$) at the same heat flows through the walls of the channel ($q_{21} = 1$). From the figure it is apparent that at $q_{21} = 1$ the temperature profile in a circular channel is unsymmetrical: the temperature of the inner wall is higher than that of the outer wall. Here the temperature maximum is located near the inner pipe. Figure 3 shows graphs of position of the temperature maximum of the fluid as a function of the ratio of the radii of the pipes forming the circular channel for various ratios of heat flows through the walls. Figure 4 gives a graph of the dimensionless difference between wall temperatures as a function of R_{21} for $q_{21} = 1$ and $q_{21} = 2$. As is apparent from these figures, the asymmetry of the temperature profile and the difference between the temperatures of the walls increase with an increase in the ratio R_{21}. The location of the maximum temperatures as functions of the ratio of heat flows through the walls are shown in Figs. 5 and 6 (the latter

* R and q with two subscripts denotes the ratio of two dimensional radii and heat flows; the first subscript pertains to the numerator, the second to the denominator.

for $R_{21} = 2$). It is apparent from these figures that the location of the temperature maximum and also the dimensionless difference between the wall temperatures of the fluid vary with q_{21}. The temperature maximum approaches the inner wall of the channel as the ratio q_{21} increases and approaches the outer wall as it decreases. In the limiting case, at $q_{21} = \infty$ (the inner wall is heat-insulated) and at $q_{21} = 0$ (the outer wall is heat-insulated), the temperature maximum will be on the inner or outer wall, respectively. For $R_{21} = 2$, the temperatures of the walls of the channel at $q_{21} = 0.785$ (Fig. 6) will be equal to one another. When the ratio q_{21} increases, the temperature of the inner pipe will become higher than that of the outer one, and their difference will increase with an increase in q_{21}. With a decrease in q_{21}, the sign of the difference between the wall temperatures will change and the temperature of the inner pipe will become lower than that of the outer one. This difference increases with a decrease in the ratio q_{21}.

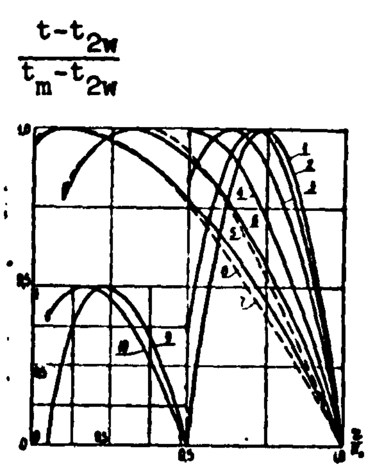

Fig. 2. Temperature distribution in circular channel with laminar flow (dotted line) and with flow with a constant cross-sectional velocity (smooth line). 1) $R_{21} = 2$, $q_{21} = 0.785$; 2) $R_{21} = 2$, $q_{21} = 1$; 3) $R_{21} = 2$, $q_{21} = 2$; 4) $R_{21} = 2$, $q_{21} = \infty$; 5, 6) $R_{21} = 10$, $q_{21} = 1$; 7, 8) $R_{21} = 100$, $q_{21} = 1$; 9) velocity distribution at $R_{21} = 10$; 10) temperature distribution at $R_{21} = 10$, $q_{21} = 1$.

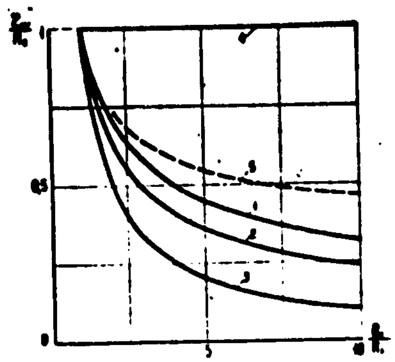

Fig. 3. Dependence of $\frac{r_m}{R_2}$ (smooth lines) and $\frac{R_m}{R_2}$ (dotted line) upon $\frac{R_2}{R_1}$.

1) $q_{21} = 1$; 2) $q_{21} = 2$;
3) $q_{21} = \infty$; 4) $q_{21} = 0$;
5) velocity maximum$\left(\frac{R_m}{R_2}\right)$.

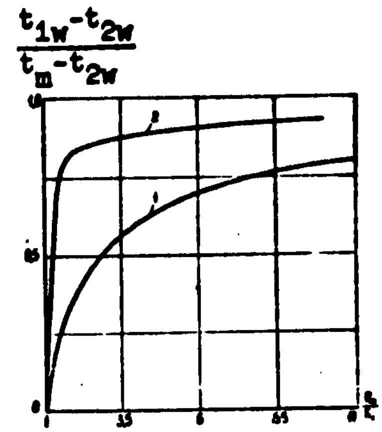

Fig. 4. Dependence of difference in wall temperatures upon $\frac{R_2}{R_1}$.

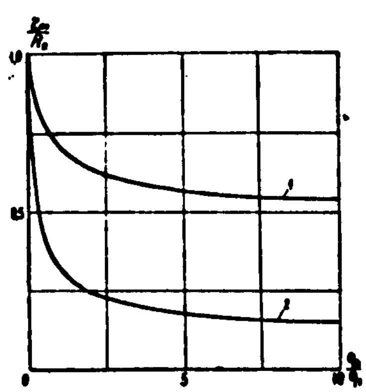

Fig. 5. Dependence of $\frac{r_m}{R_2}$ upon $\frac{q_2}{q_1}$. 1) $R_{21} = 1$; 2) $R_{21} = 10$.

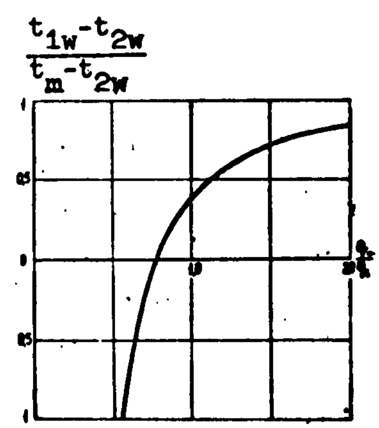

Fig. 6. Dependence of difference between wall temperatures upon $\frac{q_2}{q_1}$ for a channel with $R_{21} = 2$.

Calculation of Heat Transfer

Let us examine the calculated heat-transfer dependences for a uniform velocity distribution ($U(r) = \bar{U} =$ const). As is known, the dimensionless heat-transfer coefficient (Nusselt number) is $Nu = \frac{qd_{eq}}{\lambda \vartheta_{av}}$, where $\vartheta_{av} = t_{f\ av} - t_{w\ av}$ is the average temperature head, and d_{eq} the equivalent diameter of the channel. Together with the calculation of heat transfer for the entire channel as a whole, let us attempt to determine heat transfer separately for each of the boundary walls, in order to explain more clearly the effect of the curvature of the walls. For this let us introduce the average temperature heads, the heat-transfer coefficients, the equivalent diameters and, finally, the Nusselt number not only for the entire circular channel, but also for sections 1 and 2 separately. It is understood that this division, as before, is possible because sections 1 and 2 are separated by an adiabatic surface with radius r_m. The average temperature heads with respect to the flow rate for sections 1 and 2 are determined as:

$$\vartheta_{1\ av} = \frac{\int_{R_1}^{r_m} 2\pi \bar{U} \vartheta_1 r dr}{\pi (r_m^2 - R_1^2) \bar{U}}, \tag{11}$$

$$\vartheta_{2\ av} = \frac{\int_{r_m}^{R_2} 2\pi \bar{U} \vartheta_2 r dr}{\pi (R_2^2 - r_m^2) \bar{U}}. \tag{12}$$

Substituting here the values of ϑ_1 and ϑ_2 from (6) and (7), we obtain

$$\vartheta_{1\ av} = \frac{2(q_1 R_1 + q_2 R_2)}{\lambda (r_m^2 - R_1^2)(R_2^2 - R_1^2)} \left(\frac{r_m^4}{2} \ln \frac{r_m}{R_1} + \frac{r_m^2 R_1^2}{2} - \frac{3}{8} r_m^4 - \frac{R_1^4}{8} \right). \tag{13}$$

$$\vartheta_{2\ av} = \frac{2(q_1 R_1 + q_2 R_2)}{\lambda (R_2^2 - r_m^2)(R_2^2 - R_1^2)} \left(\frac{r_m^4}{2} \ln \frac{R_2}{r_m} - \frac{r_m^2 R_2^2}{2} + \frac{3}{8} r_m^4 + \frac{R_2^4}{8} \right). \tag{14}$$

Recalling that $q_1 = \lambda \left(\frac{d\vartheta_1}{dr}\right)_{r=R_1}, q_2 = -\lambda \left(\frac{d\vartheta_2}{dr}\right)_{r=R_2}$ and $d_{1\,eq} = 2\,(r_m - R_1)$, $d_{2\,eq} = 2\,(R_2 - r_m)$, we find the Nusselt numbers for sections 1 and 2

$$Nu_1 = \frac{q_1 d_{1eq}}{\lambda \vartheta_{1av}} = \frac{8(r_m^2 - R_1^2)^2 (r_m - R_1)}{R_1\left(4r_m^4 \ln\frac{r_m}{R_1} + 4r_m^2 R_1^2 - 3r_m^4 - R_1^4\right)}, \tag{15}$$

$$Nu_2 = \frac{q_2 d_{2eq}}{\lambda \vartheta_{2av}} = -\frac{8(R_2^2 - r_m^2)^2 (R_2 - r_m)}{R_2\left(4r_m^4 \ln\frac{R_2}{r_m} - 4r_m^2 R_2^2 + 3r_m^4 + R_2^4\right)} \tag{16}$$

The results of the numerical calculation for sections 1 and 2 at equal heat flows through the inner and outer walls for a few values of R_{21} are given in Table 1.

TABLE 1

$\frac{R_2}{R_1}$	$\frac{\lambda}{q}\,\vartheta_{1\,av}$	$\frac{\lambda}{q}\,\vartheta_{2\,av}$	Nu_1	Nu_2	$\frac{\vartheta_{2\,av}}{\vartheta_{1\,av}}$	$\frac{Nu_2}{Nu_1}$	$\frac{\alpha_2}{\alpha_1}$
2	0.126	0.187	6.51	6.29	1.48	0.96	0.67
10	0.61	2.09	7.04	6.50	3.42	0.92	0.29
100	1.59	24.52	11.31	7.34	15.42	0.65	0.065

Here α_1 and α_2 are the heat-transfer coefficients for the inner and outer walls of the circular channel.

Now let us calculate heat transfer for the entire circular channel. The average temperature head for the entire channel section let us represent in the form

$$\vartheta_{av} = \frac{\vartheta_{1av}\,(r_m^2 - R_1^2) + \vartheta_{2av}(R_2^2 - r_m^2)}{R_2^2 - R_1^2}. \tag{17}$$

Using (13), (14) and (17), let us determine the Nusselt number for

the entire channel

$$Nu_{ob} = \frac{q_{av}\, d_{eq}}{\lambda \vartheta_{av}} = \frac{8(R_2^2 - R_1^2)(R_2 - R_1)^2}{4 r_m^4 \ln R_{21} - 4 r_m^2 (R_2^2 - R_1^2) + R_2^4 - R_1^4}. \qquad (18)$$

where $q_{av} = \frac{R_1 q_1 + R_2 q_2}{R_1 + R_2}$ is the average heat flow through both walls, and $d_{eq} = 2(R_2 - R_1)$ the equivalent diameter of the circular channel.

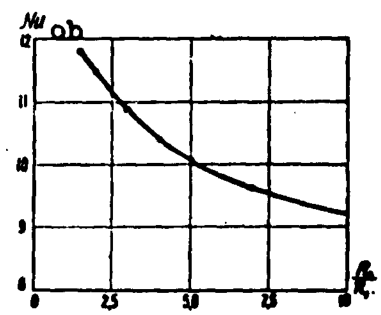

Fig. 7. The Nusselt number versus the ratio R_2/R_1.

Figure 7 shows the Nusselt number as a function of the ratio of the radii of the outer and inner pipes. As is apparent from the figure, in the limiting case of transition to a plane channel ($R_{21} = 1$), the Nusselt number becomes equal to about 12. In the other limiting case — transition to a circular pipe ($R_{21} \to \infty$)— the value of Nu_{ob} approaches 8. Both these limiting cases, naturally, coincide with the results of the corresponding solutions given in the literature [1].

The Effect of the Velocity Distribution

In order to evaluate the effect of the velocity distribution along the cross section of the channel, which was not taken into account earlier, let us examine the solution for laminar flow in a circular channel. Let us limit ourselves to the particular case of equality of heat flows through the walls of the channel ($q_{21} = 1$). The velocity distribution of laminar flow in a circular channel under isothermal conditions, as is known, satisfies the equation [2]

$$U(r) = \frac{\Delta p}{4\eta l}\left(R_2^2 - r^2 + \frac{R_2^2 - R_1^2}{\ln R_{21}} \cdot \ln \frac{r}{R_2}\right). \qquad (19)$$

The value of the radius R_m corresponding to the maximum velocity calculated by this equation has the form

$$R_m = \left(\frac{R_2^2 - R_1^2}{2 \ln R_{21}} \right)^{\frac{1}{2}}. \tag{20}$$

The velocity profiles constructed according to Formula (19) are unsymmetrical. The velocity maximum does not coincide with the middle of the circular channel and, as the temperature maximum in the previous calculation at $q_{21} = 1$, it is situated near the inner pipe.

The average flow rate of the fluid is determined from the equation

$$\bar{U} = \frac{\int_{R_1}^{R_2} 2\pi U(r) r dr}{\pi (R_2^2 - R_1^2)} = \frac{\Delta p}{8 \eta l} \left(R_2^2 - R_1^2 - \frac{R_2^2 - R_1^2}{2 \ln R_{21}} \right). \tag{21}$$

Substituting the value of $U(r)$ from (19) and of $\bar{U}$ from (21) into (5), we obtain an equation for determining the radius of the maximum of the temperature profile

$$\frac{r_m^2}{R_1^2} \left(2 R_{21}^2 - \frac{R_{21}^2 - 1}{\ln R_{21}} + \frac{R_{21}^2 - 1}{\ln R_{21}} \cdot \ln \frac{r_m}{R_2} - \frac{r_m^2}{R_2^2} R_{21}^2 \right) = R_{21} + R_{12} - \frac{R_{21} - R_{12}}{\ln R_{21}}. \tag{22}$$

The graphical solution of this equation for r_m shows that the locations of the temperature maximum for laminar flow of a fluid and for flow with a constant cross-sectional velocity coincided, for all practical purposes. As an example, Table 2 gives values of the radii of the maximum of the temperature profile for various values of the ratio R_{21} at $q_{21} = 1$. Values of the maximum velocity profile for laminar flow are given here for comparison.

It should be noted that the radius of the maximum of the temperature profile r_m for any value of R_{21} is less than the radius of the maximum of the velocity profile R_m, as determined by Eq. (20) (in Fig. 3 the dotted curve shows R_m/R_2 as a function of R_{21}).

TABLE 2

	$\frac{R_2}{R_1}$	2	3	4	5	6	7	8	9	10
Laminar flow	$\frac{R_m}{R_2}$	0.737	0.640	0.585	0.542	0.518	0.500	0.481	0.470	0.460
Laminar flow	$\frac{r_m}{R_2}$	0.701	0.570	0.500	0.447	0.406	0.377	0.352	0.330	0.313
Flow with constant cross-sectional velocity	$\frac{r_m}{R_2}$	0.707	0.577	0.502	0.448	0.408	0.378	0.354	0.333	0.316

For clarity, the velocity and temperature distribution for laminar flow in a circular channel with $R_{21} = 10$ and $q_{21} = 1$ is given in the corner of Fig. 2.

The solution of Eq. (3) taking (19) and (20) into account gives the cross-sectional temperature distribution for a circular channel in the form:

$$\frac{t - t_{2w}}{t_m - t_{2w}} = \frac{\left[A + B\left(D + \frac{r^2}{4}\right)\right] \ln \frac{R_2}{r} + \frac{3R_2^4 + r^4}{16} - \frac{B}{4}(R_2^2 - r^2) - \frac{R_2^2 r^2}{4}}{\left[A + B\left(D + \frac{r_m^2}{4}\right)\right] \ln \frac{R_2}{r_m} + \frac{3R_2^4 + r_m^4}{16} - \frac{B}{4}(R_2^2 - r_m^2) - \frac{R_2^2 r_m^2}{4}}, \tag{23}$$

where

$$A = \frac{r_m^4}{4} - \frac{R_2^2 r_m^2}{2}, \quad B = \frac{R_2^2 - R_1^2}{\ln R_{21}}, \quad D = \frac{r_m^2}{4}\left(2 \ln \frac{R_2}{r_m} + 1\right).$$

Curves of the temperature distribution constructed according to Eq. (23) for various values of R_{21} are given in Fig. 2 by dotted lines. As is apparent from this figure, the curves of the temperature distribution for laminar flow and for flow with constant cross-sectional velocity differ from one another very little with regard to the location of the temperature maximum. This allows it to be assumed that the location of the temperature maximum for a turbulent velocity profile, without taking into account the effect of turbulent thermal conductivity, remains practically the same. This result pertains to the practically important case of fluids with a very low Prandtl number (liquid metals).

This conclusion cannot be extended to the value of the temperature gradient near the walls of a channel and, therefore, to heat transfer. As is apparent from Fig. 2, near both walls the temperature gradient in the case of $U(r) = \overline{U} = \text{const}$ is markedly higher than when the laminar velocity profile is taken into account. This effect of the velocity profile upon heat transfer is in qualitative agreement

with that known for heat transfer in a circular pipe and in a plane channel. Let us remember that in the first case ($R_{21} = \infty$, circular pipe) the ratio of the Nusselt numbers for a constant velocity and parabolic distribution equals ~ 1.85, while in the second case ($R_{21} = 1$, plane channel) it equals ~ 1.45. These data can be used for an approximate evaluation of the effect of the velocity profile upon heat transfer in a circular channel. When necessary, an accurate calculation can be made similarly to the previoue one, but it is not given here owing to its awkwardness. Let us note only, for example, that at $R_{21} = 2$ and 10 and $q_{21} = 1$ the ratio of Nusselt numbers calculated with and without taking the velocity distribution into account are 1.55 and 1.65 respectively.

Conclusions

As can be seen from the calculation made, the geometry of the channel has the main effect on the temperature field and heat transfer, just as on the flow itself. The asymmetry of the temperature profile, particularly the location of the temperature maximum, is a function of the ratio of the radii. The same sort of dependence is characteristic of the Nusselt number in stabilized heat transfer, which in general is determined by two parameters: the ratio of the radii and the ratio of the heat flows through walls of the channel.

The determination of heat transfer separately for the inner and outer walls of a circular channel shows that for the inner wall it is more intensive, especially at high ratios of the radii of the outer and inner pipes of the circular channel.

The calculations (for the case of constancy of heat flows through the walls of the channel) show that the location of the

temperature maximum is slightly dependent upon the velocity profile. Regarding the effect of the latter on heat transfer, it apparently has the same character as for heat transfer in the two limiting cases (in a circular pipe and in a plane channel).

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